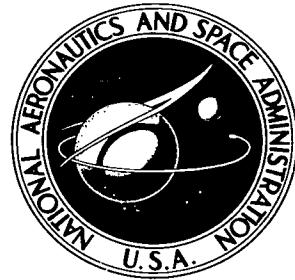


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## DUAL-ACTION GAS THRUST BEARING FOR IMPROVING LOAD CAPACITY

*Izhak Etsion*

*Lewis Research Center  
Cleveland, Ohio 44135*



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16. Abstract <p>The principle of utilizing hydrodynamic effects in diverging films to improve the load-carrying capacity in gas thrust bearings is discussed. A new concept of a dual-action bearing based on that principle is described and analyzed. The potential of the new bearing is demonstrated both analytically for an infinitely long slider and by numerical solution for a flat sector-shaped thrust bearing. It is shown that the dual-action bearing can extend substantially the range of load-carrying capacity in gas-lubricated thrust bearings and can improve their efficiency.</p>			
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## DUAL-ACTION GAS THRUST BEARING FOR IMPROVING LOAD CAPACITY

by Izhak Etsion\*

Lewis Research Center

### SUMMARY

The principle of utilizing hydrodynamic effects in diverging films to improve the load-carrying capacity in gas thrust bearings is described. By placing a stationary member with a diverging film next to the so-far inactive surface of the runner, the pressure difference across the runner is increased and consequently more load is obtained from a given bearing area.

The new concept, denoted as a dual-action bearing, is analyzed and its potential demonstrated by solving analytically a simple case of an infinitely long slider. Load capacity improvement of up to 100 percent is shown to be possible.

A numerical solution is presented for a dual-action bearing consisting of flat sector-shaped pads. A wide range of compressibility is covered for a pad radius ratio of 0.5 and a sector angle of 45°. The bearing eccentricity is defined, and the procedure for calculating bearing performance is outlined. Results are given for the case of zero eccentricity. It is shown that the improvement in load-carrying capacity is more than 90 percent over the whole range of compressibility.

Although some stability problems might occur under conditions of low load and high speed, the dual-action bearing could be very useful in increasing load-carrying capacity and improving efficiency.

### INTRODUCTION

The increase in the speed and operating temperature of rotating machinery in recent years has made the use of gas-lubricated bearings attractive in many applications. An inherent weakness of self-acting gas bearings is their low load-carrying

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\* National Research Council - National Aeronautics and Space Administration  
Research Associate.

capacity, which is no more than 1 or 2 atmospheres. Therefore, whenever high or even moderate loads are considered, external pressurization and/or large bearing area are needed. However, pressurization complicates the design, and bearing size is usually constrained. It is therefore natural to search for designs that extract the maximum load from a given bearing size. Several such designs have been offered and are now commonly used. Among them are stepped bearings, tilting pads, and various grooved configurations (ref. 1).

The load-carrying capacity in hydrodynamic bearings is developed by pressure generated within a lubricant film due to the wedge action between varying gaps of two surfaces. To be more specific the load is supported by the pressure difference across the loaded member of the bearing; and the higher that difference, the higher the load-carrying capacity.

The pressure difference across the loaded member of a bearing can be increased by increasing the pressure in the high-pressure region or by decreasing it in the low-pressure region. Much attention has been paid to the first possibility, but the second has so far been neglected. This approach is justified in the case of liquid lubricants, where the maximum reduction in the low-pressure region can be only 1 atmosphere, which is two orders of magnitude lower than the common pressure increase. However, with gas-lubricated bearings the reduction might be of the same order as the pressure increase and could therefore contribute substantially to the bearing load-carrying capacity.

Referring to bearings that develop their load-carrying capacity merely by a pressure increase on one side of the loaded member as single-action bearings, we shall use the term dual-action bearings in those cases where the load-carrying capacity is the result of both a pressure increase on one side and a pressure decrease on the other side of the loaded member.

In reference 2 the idea of improving load-carrying capacity by pressure reduction is applied to a gas journal bearing. Although journal bearings, because of their converging-diverging film shape, are inherently dual-action bearings, it is shown in reference 2 that a proper design can further improve the load-carrying capacity. This was accomplished by placing a forward-facing step in the high-pressure region and a backward-facing step in the low (subambient)-pressure region (fig. 1), thus increasing the pressure difference across the loaded journal.

In contrast to journal bearings, all conventional hydrodynamic thrust bearings in present use are of the single-action type, having a relatively large inactive area. Therefore it seems that they can be improved more than can journal bearings from proper design based on the dual-action principle.

In the following sections a dual-action gas thrust bearing is described and a flat sector-pad configuration analyzed to demonstrate the potential of the dual-action principle.

## THE DUAL-ACTION GAS THRUST BEARING

It is well known that the generation of above-ambient pressure in a lubricating film is related to surface movement in the direction of the film convergence. When the surface moves in the direction of film divergence, the result is subambient pressure. A single-action thrust bearing consists of a runner that rotates next to a stationary member (sector pad or grooved disk), so that the rotation is in the direction of film convergence (fig. 2(a)). By placing an identical stationary member next to the other surface of the runner, rotation in the direction of film divergence of that member is obtained. Hence, pressures below ambient are generated on that surface and the bearing becomes a dual-action one (fig. 2(b)).

The load carried by the dual-action bearing is the sum of that obtained from the basic single-action configuration and the additional load obtained from the diverging-film portion of the bearing. Since the performance characteristics of many single-action configurations are already known, it is sufficient to solve only for the diverging films and to find their contribution.

Before a complete solution of the two-dimensional Reynolds equation is performed, it is worthwhile to examine the potential of the dual-action bearing for a simple case. This will be done now for an infinitely long slider, where the Reynolds equation is easily solved at both extremes of the velocity range (e.g., ref. 3). Consider the one-dimensional Reynolds equation

$$\frac{d}{dx} \left( ph^3 \frac{dp}{dx} \right) = 6\mu U \frac{d(ph)}{dx} \quad (1)$$

This equation integrates to

$$\frac{dp}{dx} = \frac{6\mu U}{ph^3} (ph - C_1) \quad (2)$$

where  $C_1$  is a constant of integration.

At very high speeds as  $U \rightarrow \infty$ ,  $dp/dx$  can remain finite only if  $ph \rightarrow C_1$ . This gives the high-speed asymptote for the gas-lubricated bearing. Hence, for the slider of figure 2(a) we have

$$p = p_a \frac{h_M}{h}$$

where  $p_a$  is the ambient pressure at the inlet clearance  $h_M$  and

$$h = h_M - \alpha x$$

The load per unit length of that slider at very high speed is

$$\left(\frac{W}{L}\right)_c = \int_0^B p_a \left(\frac{h_M}{h} - 1\right) dx = \frac{p_a h_M}{\alpha} \left(\ln k - \frac{k-1}{k}\right) \quad (3)$$

where  $k = h_M/h_m$ .

Reversing now the pad of figure 2(a) so that its leading and trailing edges are interchanged, the pressure distribution becomes

$$p = p_a \frac{h_m}{h}$$

where now  $h = h_m + \alpha x$ .

The load per unit length contributed by the diverging film is

$$\left(\frac{W}{L}\right)_d = \int_0^B p_a \left(1 - \frac{h_m}{h}\right) dx = \frac{p_a h_M}{\alpha} \left(\frac{k-1}{k} - \frac{1}{k} \ln k\right) \quad (4)$$

Dividing equation (4) by equation (3) gives the ratio between the loads contributed by the two configurations. Thus,

$$\eta = \frac{W_d}{W_c} = \frac{k-1-\ln k}{k \ln k - k + 1} \quad (5)$$

Differentiating equation (5) with respect to  $k$ , we find that  $\eta_{\max} \rightarrow 1$  as  $k \rightarrow 1$ , which means that the dual-action bearing is most effective at low loads.

Figure 3 presents values of  $\eta$  for various values of  $k$ . It is clear from the figure that the improvement in load-carrying capacity is substantial over a wide range of loads. Different values of  $k$  could be assigned to the converging and diverging configurations. However, it can be shown that, if the minimum film thickness is to be kept the same, the value of  $k$  that maximizes the pressures in a converging film will also minimize the subambient pressures in the case of a diverging film. This will

maximize the total load of the dual-action bearing of figure 2(b).

At very low speed, equation (1) becomes (ref. 3)

$$\frac{d}{dx} \left( h^3 \frac{dp}{dx} \right) = 6\mu U \frac{dh}{dx} \quad (6)$$

Integrating twice gives for the pressure

$$p_c - p_a = 6\mu U \int_0^x \frac{h - C_2}{h^3} dx \quad (7)$$

where  $C_2$  is a constant of integration. The load per unit length is the integral from  $x = 0$  to  $x = B$  of the right side of equation (7). Now reversing the direction of  $U$  in figure 2(a) results in changing the sign of  $U$  in equation (7) and the pressure distribution becomes

$$p_a - p_d = 6\mu U \int_0^x \frac{h - C_2}{h^3} dx \quad (8)$$

Again if we choose identical clearances on both sides of the runner, it is clear from equations (7) and (8) that the load contributed by the diverging portion of the dual-action bearing is equal to that contributed by the converging portion. Note that  $h$  has not been specified; hence, the analysis holds for any arbitrary clearance. The conclusion is that at very low speed we always have  $\eta = 1$ , which means 100 percent improvement over the single-action bearing. So far we have been dealing with bearings where the runner is at the middle of the gap between the top and bottom pads of figure 2(b). To determine completely the bearing performance, its behavior on both sides of the mid-plane (fig. 4) should be known.

Having a dual-action bearing with a net clearance  $2C$  and a runner displaced a distance  $e$  from the midplane, we shall define the bearing eccentricity  $\epsilon$  as the ratio  $e/C$ , where positive eccentricity is in the direction of loading. From this definition it is clear that the runner can assume any eccentricity  $\epsilon$  between -1 and 1. Denoting by subscripts  $c$  and  $d$  the converging film and diverging film pads, respectively, we have from figure 4

$$(h_m)_c = C - e = C(1 - \epsilon) \quad (9a)$$

$$(h_m)_d = C + e = C(1 + \epsilon) \quad (9b)$$

For identical pads we have  $h_M = h_m + \alpha B$ ; hence

$$k_c = 1 + \frac{\alpha B}{C(1 - \epsilon)} \quad (10a)$$

$$k_d = 1 + \frac{\alpha B}{C(1 + \epsilon)} \quad (10b)$$

Substituting into equations (3) and (4) and adding them up gives the total load of the dual-action bearing at any eccentricity  $-1 < \epsilon < 1$ ,

$$\frac{W}{L} = \left( \frac{p_a C}{\alpha} \right) \left\{ (1 - \epsilon) \left[ 1 + \frac{\alpha B}{C(1 - \epsilon)} \right] \ln \left[ 1 + \frac{\alpha B}{C(1 - \epsilon)} \right] - (1 + \epsilon) \ln \left[ 1 + \frac{\alpha B}{C(1 + \epsilon)} \right] \right\} \quad (11)$$

Differentiating  $W$  in equation (11) with respect to  $\epsilon$ , we have for the bearing stiffness

$$K = \frac{p_a B L}{\psi} \left\{ \frac{\psi}{(1 - \epsilon)} - \ln \left[ 1 + \frac{\psi}{(1 - \epsilon)} \right] + \frac{\psi}{\psi + (1 + \epsilon)} - \ln \left[ 1 + \frac{\psi}{(1 + \epsilon)} \right] \right\} \quad (12)$$

where  $\psi = \alpha B / C$ .

Values of  $K$  are plotted in figure 5. It is seen that the stiffness  $K$  decreases with  $\epsilon$  and that there is a critical eccentricity  $\epsilon_{cr}(\psi)$  where the stiffness vanishes. For  $\epsilon < \epsilon_{cr}$  the stiffness is negative and the bearing fails. A similar result is obtained for the low-speed range.

In a way, such behavior of a dual-action bearing is similar to instability problems in lightly loaded gas journal bearings.

#### NUMERICAL SOLUTION FOR FINITE BEARING

To demonstrate the potential of a realistic dual-action thrust bearing, a flat sector-shaped pad was analyzed. Both the pitch and roll of the pad were taken into account, and an accurate film thickness variation was used. The geometry of the sector pad is shown in figure 6, and a complete analysis along with the results for converging configurations is given in reference 4. The Reynolds equation in nondimensional form as derived in reference 4 is

$$\frac{\partial^2 Q}{\partial R^2} + \frac{1}{R^2} \frac{\partial^2 Q}{\partial \theta^2} - \left[ \frac{\partial}{\partial R} \left( \ln \frac{H}{R} \right) \right] \frac{\partial Q}{\partial R} - \left[ \frac{1}{R^2} \frac{\partial}{\partial \theta} \left( \ln \frac{H}{R} \right) \right] \frac{\partial Q}{\partial \theta} = \frac{\Lambda h_m^2}{H \sqrt{Q}} \frac{\partial Q}{\partial \theta} \quad (13)$$

where  $Q = (PH)^2$ ;  $P$  is the nondimensional pressure defined as  $P = p/p_a$ ; and  $\Lambda$ , the compressibility number, is given by  $\Lambda = 6\mu\omega r_o^2/p_a h_m^2$ . It is shown in reference 4 that a combined pitch and roll about some point can be transformed into a pure pitch  $\gamma$  about a certain radial line at an angle  $\theta_p$ . The nondimensional film thickness then becomes

$$H = 1 + \frac{\gamma r_o}{h_p} R \sin(\theta_p - \theta) \quad (14)$$

Equation (13) is solved in reference 4 for positive values of  $\Lambda$  ranging from 1 to 100, thus covering a wide range of speeds for a runner rotating in the direction of decreasing film thickness. Solving equation (13) for negative values of  $\Lambda$  provides the results for the case of rotation in the opposite direction, namely in the direction of increasing film thickness. Also in reference 4 various values of  $\theta_p/\beta$  were examined, and it was found that the optimum value of  $\theta_p/\beta$  for maximizing load-carrying capacity is 1, in which case  $h_p = h_m$ . For the present example, a pad having a sector angle  $\beta$  of  $45^\circ$  and a radius ratio  $r_i/r_o$  of 0.5 was selected. Equation (13) was solved for compressibility numbers  $\Lambda$  ranging from -1 to -100 at various  $\theta_p/\beta$  values. This was done to determine the optimum  $\theta_p$ . Again, as in the case of converging films, it was found that a  $\theta_p/\beta$  of 1 maximizes the load-carrying capacity over the whole range of compressibility.

Figure 7 presents the unit load for a diverging film sector tilted about its leading edge. The results are similar to those shown in figure 8 (from ref. 4) for a converging film sector tilted about its trailing edge.

To obtain the load-carrying capacity of the dual-action bearing at a given eccentricity  $\epsilon$ , compressibility  $\Lambda = 6\mu\omega r_o^2/p_a C^2$ , and clearance parameter  $\varphi = \gamma r_o/C$ , the individual compressibility and clearance parameters of each of the sectors have to be determined. Using equations (9a) and (9b) we have for the converging film sector

$$\Lambda_c = \frac{6\mu\omega r_o^2}{p_a (h_m^2)_c} = \frac{\Lambda}{(1 - \epsilon)^2}$$

$$\varphi_c = \frac{\gamma r_o}{(h_m)_c} = \frac{\varphi}{(1 - \epsilon)}$$

Similarly, for the diverging film sector,

$$\Lambda_d = \frac{\Lambda}{(1 + \epsilon)^2}$$

$$\varphi_d = \frac{\varphi}{(1 + \epsilon)}$$

Design curves for any  $\Lambda$ ,  $\gamma r_0/C$ , and  $\epsilon$  can now be plotted. This is done by adding the unit loads  $W_c/p_a A$  and  $W_d/p_a A$  obtained from figures 7 and 8, at the appropriate values of  $\Lambda_c$ ,  $\varphi_c$  and  $\Lambda_d$ ,  $\varphi_d$ , respectively.

Figure 9 presents the unit load for a centrally positioned dual-action bearing, namely at  $\epsilon = 0$ . In this particular case we have  $\Lambda_c = \Lambda_d = \Lambda$ , and  $\varphi_c = \varphi_d = \varphi$ . It is seen that, similar to the single-action bearing (fig. 8), the unit load reaches a maximum at clearance parameters between 2 and 5, depending on the compressibility  $\Lambda$ . In other words, for any given speed  $\omega$  and clearance  $C$  (or minimum film thickness  $h_m$  in the case of a single-action bearing), there is a maximum value to the load that can be carried by the bearing. These maximum values for both the single-action and the centrally positioned dual-action bearings are plotted in figure 10 against the compressibility number  $\Lambda$ . It can be seen that over the whole range, up to  $\Lambda = 100$ , the improvement in maximum load-carrying capacity with the dual-action bearing is more than 90 percent over the single-action bearing. The conclusion is that at any given operating conditions of speed and minimum film thickness, the dual-action bearing is able to carry more load than the single action. Alternatively, for a given speed and load the dual-action bearing operates at a lower compressibility number. Hence, the minimum clearance needed to carry the load is increased and the danger of surface rub is reduced; or the outer radius can be decreased, resulting in lower weight.

In reference 4 it is shown that the efficiency of a bearing, as expressed by the ratio of load carried to power lost, decreases when  $\Lambda$  increases. Hence, the dual-action bearing, by operating at lower  $\Lambda$ , offers greater efficiency than the single-action bearing. This advantage and the ability to extend the range of load-carrying capacity beyond the present limits make the dual-action bearing an attractive design.

#### CONCLUDING REMARKS

A new concept for a gas thrust bearing, denoted as a dual-action bearing, was presented. Its potential of improving the load-carrying capacity of gas thrust bearings was demonstrated both analytically for an infinitely long slider and through a numerical solution of a sector-shaped thrust bearing. For a given load and speed, the dual-action bearing should consume less power than the conventional single-action bearing,

thus offering greater efficiency.

At light loads and high speeds, when the eccentricity becomes negative, stability problems may occur. The suction on the low-pressure side may increase faster than the pressure decrease on the high-pressure side of the runner, causing the runner to contact the diverging film pad. It is believed, however, that this problem can be solved by controlling the bearing clearance or by providing an auxiliary device like a carbon button at the end of the shaft to limit its axial movement and prevent the runner from contacting the diverging pad. Such a device also provides a means of carrying a momentary thrust load in a reverse direction.

At moderate and high loads, however, the dual-action bearing might be very useful in extending the load-carrying range of gas bearings and in increasing their efficiency.

The design is not limited to flat sector pads; other configurations such as stepped pads and grooved disks can be used as the stationary member in a dual-action thrust bearing.

Lewis Research Center,  
National Aeronautics and Space Administration,  
Cleveland, Ohio, April 27, 1976,  
505-04.

## APPENDIX - SYMBOLS

A	pad area, $\beta(r_o^2 - r_i^2)/2$
B	slider width in direction of motion
C	bearing clearance
$C_1, C_2$	constants of integration
e	runner displacement from bearing midplane
H	nondimensional film thickness, eq. (14)
h	film thickness
$h_M$	maximum film thickness
$h_m$	minimum film thickness
K	bearing stiffness, $dW/d\epsilon$
k	clearance ratio, $h_M/h_m$
L	slider length perpendicular to direction of motion
P	nondimensional pressure, $p/p_a$
p	pressure
$p_a$	ambient pressure
Q	$(PH)^2$
R	nondimensional radius, $r/r_o$
r	radial coordinate
$r_i$	inner radius of pad
$r_o$	outer radius of pad
t	runner thickness
U	linear velocity
W	load
x	coordinate
$\alpha$	slider slope
$\beta$	angular extent of pad (sector angle)

$\gamma$	pad tilt
$\epsilon$	bearing eccentricity, $e/C$
$\eta$	load ratio, $W_d/W_c$
$\theta$	angular coordinate
$\Lambda$	compressibility number, $6\mu\omega r_o^2/p_a C^2$ or $6\mu\omega r_o^2/p_a h_m^2$
$\mu$	dynamic viscosity
$\varphi$	sector-pad clearance parameter, $\gamma r_o/C$ or $\gamma r_o/h_m$
$\psi$	slider clearance parameter, $\alpha B/C$
$\omega$	shaft rotational speed

Subscripts:

c	film converging in direction of motion
cr	critical
d	film diverging in direction of motion
p	radial pivot line
max	maximum

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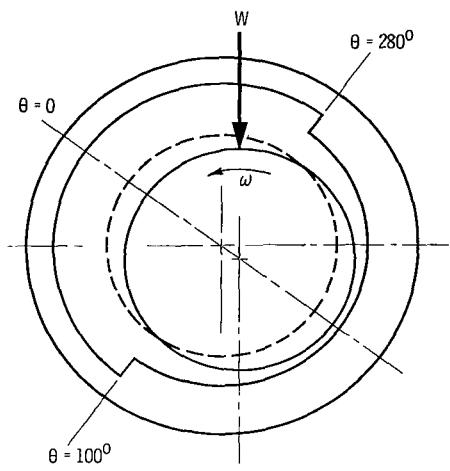


Figure 1. - A doubly stepped journal bearing.  
(From ref. 2.)

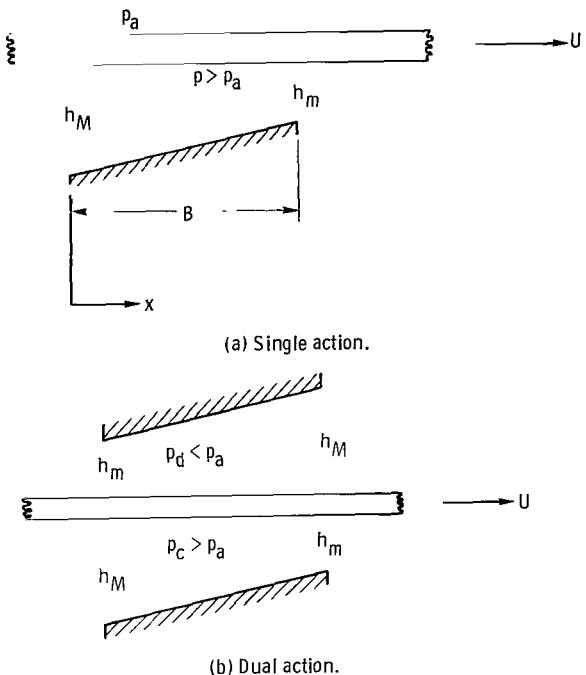


Figure 2. - Typical thrust bearings.

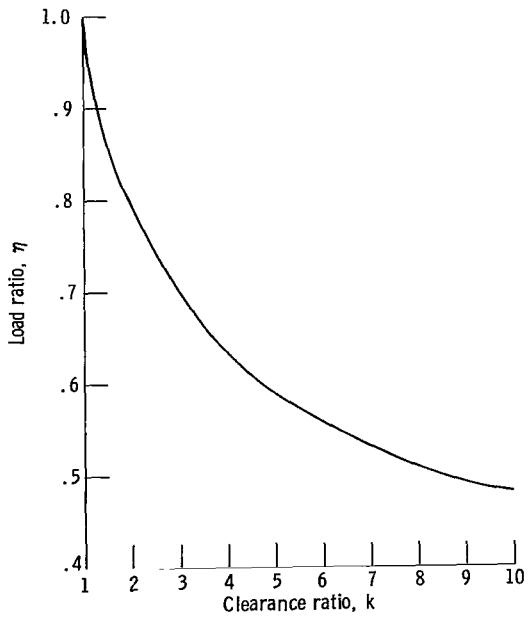


Figure 3. - Load-capacity improvement for an infinitely long dual-action slider.

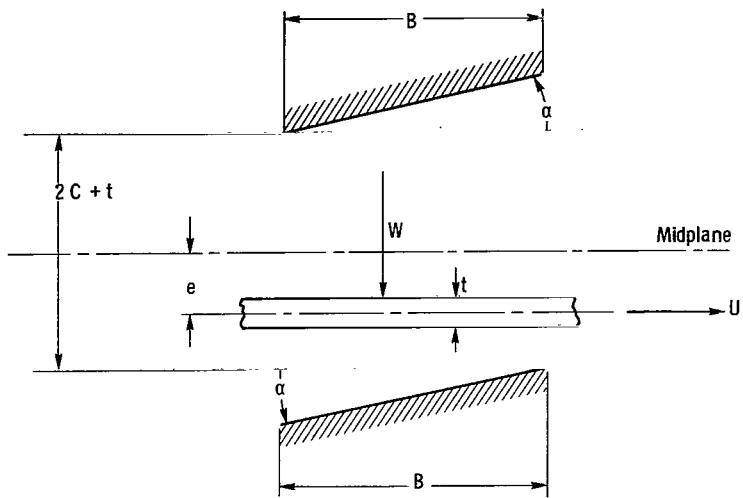


Figure 4. - Eccentric dual-action thrust bearing.

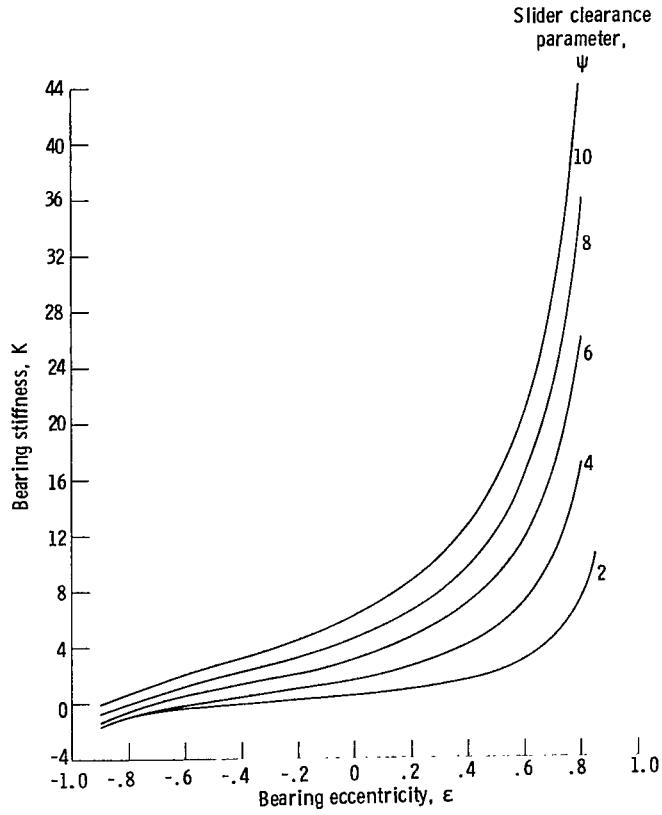


Figure 5. - Stiffness of infinitely long dual-action slider.

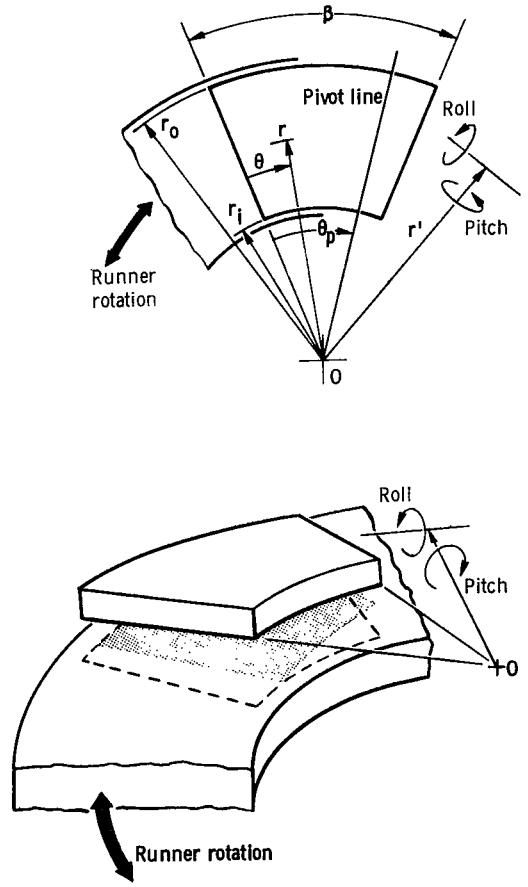


Figure 6. - Geometry of sector pad.

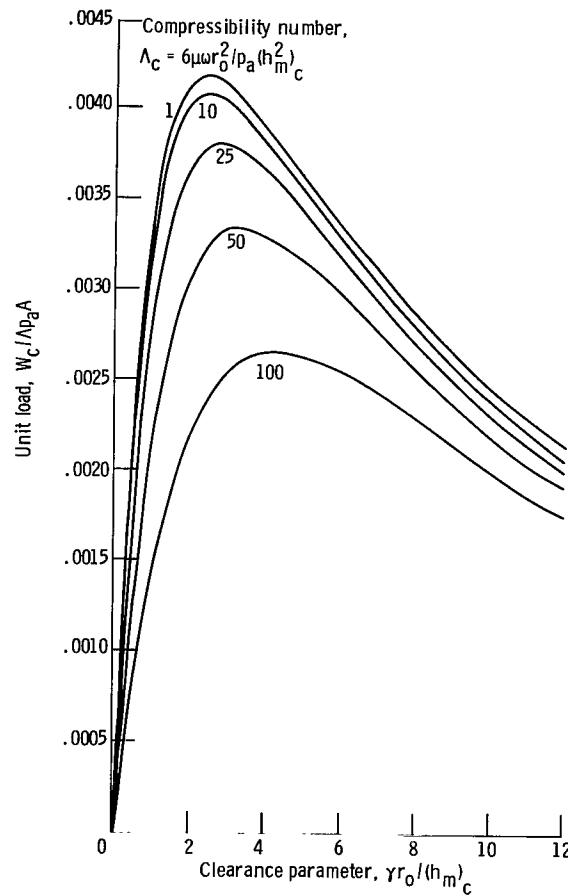


Figure 8. - Unit load of optimally tilted, converging-film sector. Ratio of inner to outer pad radius,  $r_i/r_0$ , 0.5; angular extent of pad,  $\beta$ ,  $45^\circ$ . (From ref. 4.)

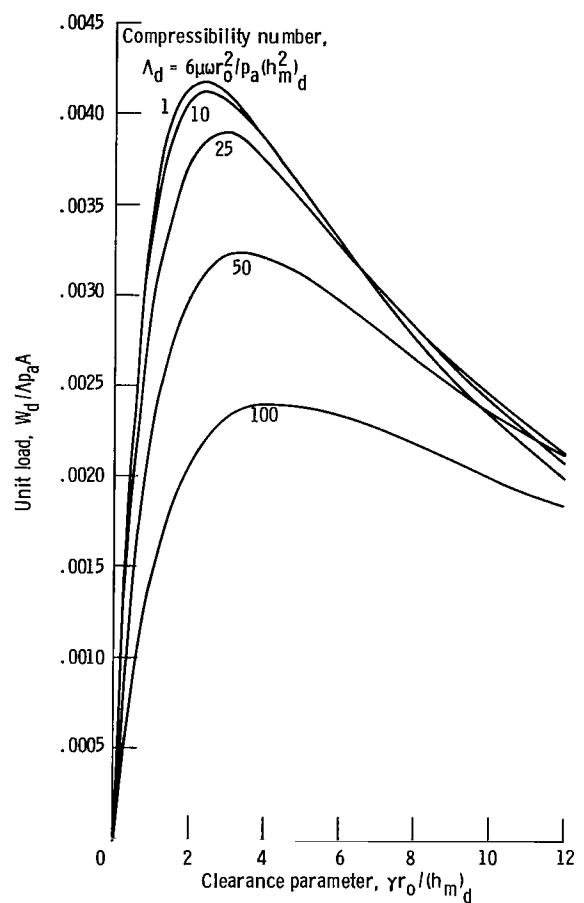


Figure 7. - Unit load of optimally tilted, diverging-film sector. Ratio of inner to outer pad radius,  $r_i/r_0$ , 0.5; angular extent of pad,  $\beta$ ,  $45^\circ$ .

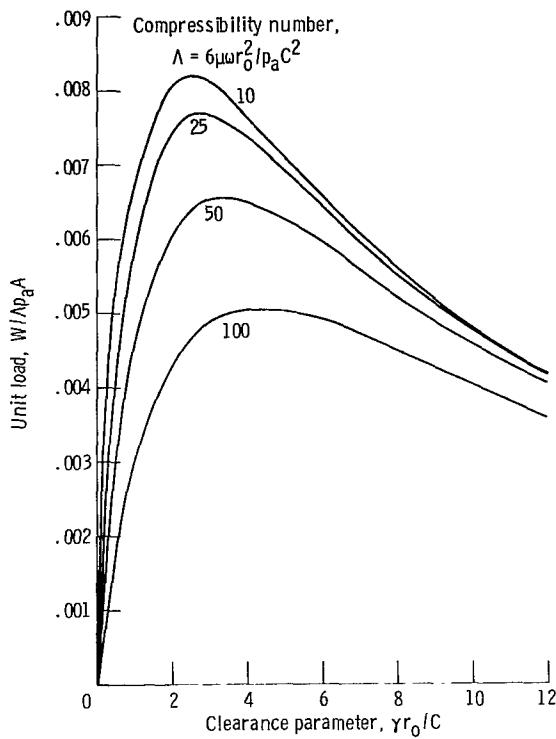


Figure 9. - Unit load of dual-action, sector-shaped thrust bearing. Eccentricity,  $\epsilon$ , 0; ratio of inner to outer pad radius,  $r_i/r_o$ , 0.5; angular extent of pad,  $\beta$ ,  $45^\circ$ .

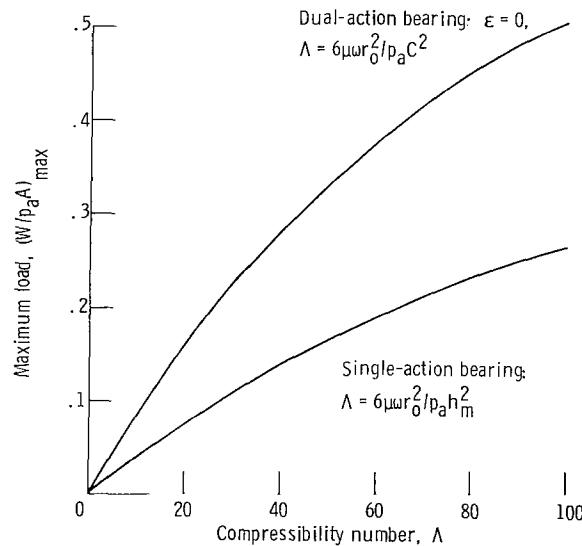


Figure 10. - Maximum load capacity of sector-shaped thrust bearing. Ratio of inner to outer pad radius,  $r_i/r_o$ , 0.5; angular extent of pad,  $\beta$ ,  $45^\circ$ .

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